HIGH-SPEED GEAR VIBRATION AND NOISE EXPERIENCE

by

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ABSTRACT

High-speed gear pinion precession-shuttling and noise, caused by overhanging weight of the coupling and by misalignment forces generated by tooth friction in the coupling, are being controlled by reduced bearing clearances. A statistical analysis of high-speed gear, mesh-frequency, vibrations was used to decide that sleeve bearings would not control the precession-shuttling adequately and that use ot tilting-shoe radial bearings was necessary.

Noise data collected to solve the pinion precession-shuttling problem indicate that acoustic passages in the gear-box may be excited by windage at frequencies not related to vibration multiples of mechanical running speed or gear mesh-frequency. Measured resonant frequencies of structural sections of the gear-box when matched to measured acoustic frequencies or mechanical vibration frequencies also appear to cause unnecessary noise.

These experiences suggest that some relatively minor design changes, by gear manufacturer-customer agreement, could reduce the chance of mechanical malfunction by installation of tilting-shoe bearings and could eliminate noise sources in preference to use of noise abating gear-box covers.

INTRODUCTION

The objective of this paper is to describe a gear-box pinion precession-shuttling problem in enough detail for other gear users to:

1. Recognize the problem.
2. Be able to evaluate the severity.
3. Proceed confidently with corrective repair.
4. Realize the indirect benefits of corrective action.
5. Purchase gears free of precession-shuttling problems or purchase gear-boxes readily adaptable to corrective modification.

The shaft motion describing precession is conical with a node at a center of gravity or a center of moments determined by weight distribution, unbalance distribution and power transmission forces. The shuttling is an axial motion limited only by backlash and bearing clearances. The conical motion and the axial motion are superimposed on each other to make the motion we call "precession-shuttling."

The gear to be discussed was put in service in 1970 as part of the compression train shown in Figure 1. A 6000-horsepower, 1780 rpm motor drives two 9302 rpm, 10-stage, double-case compressors through the speed increasing gear shown in Figure 2. Gear data is shown in Table 1.

The data tabulated in Table 1 does not represent any design frontiers; however, the gear-box appears to have a relatively high pitch line velocity (21,318 feet per minute), a high tooth loading (854 pounds per inch of tooth face), and large teeth (1.12 teeth per inch) when compared to 16 other high-speed gears in service. The tooth finish is excellent; corrective pitting of the tooth face is nonexistent; the static and dynamic tooth contact pattern is excellent; and there is virtually no tooth wear.

GEAR START-UP

In late 1970 this compressor train was being field checked prior to plant operation. The established procedure was to operate the driver solo, then couple and operate each unit of the compressor train in sequence to assure mechanical integrity of each piece of the train and finally the system including lube system, seal system, and instruments. At this time, the motor and gear had been alignment checked, coupled, and a solo plate had been placed on the high-speed pinion coupling half to make ready for gear-box operation.

When the motor and gear-box were started together, they ran beautifully for about one minute before pinion precession-shuttling with its very disturbing noise and vibration started.

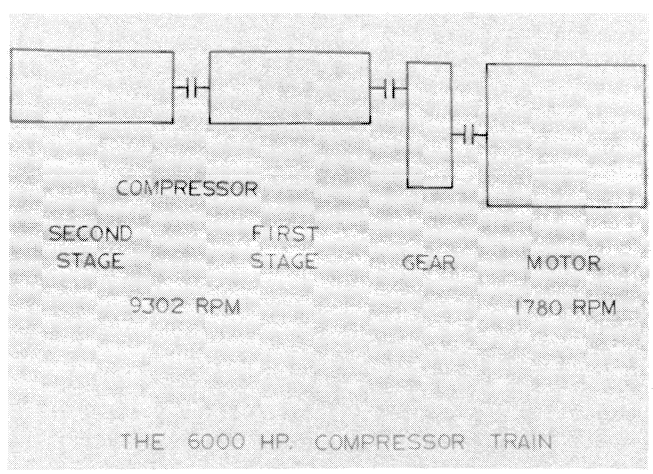

Figure 1. Block Diagram of Motor, Gear, and Compressor Train.

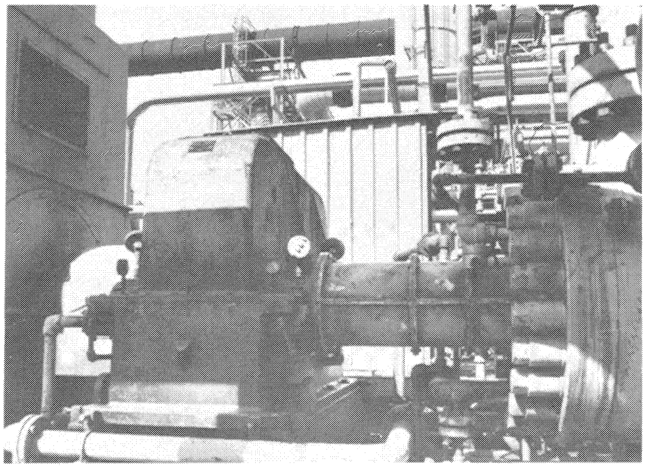

Figure 2. Gear Box Subject to Precession-Shuttling of Pinion.

Fortunately, the vibration and noise severity increased gradually as the oil warmed which allowed operation for another 30 seconds before shutdown action was required. After shutdown, the gear and pinion teeth were inspected through the hand hole; the bearings were examined by removing labyrinths and shaft end covers for visual access; and the bearing clearance was roughly measured with dial indicators by prying up the shafts. Since there was no apparent damage to the gears or bearings, a second trial run was made to collect diagnostic data. Vibration velocity transducers were attached to the bearing housings and the output of the transducers was tape recorded while the gear-box was started and operated for about 90 seconds, 30 seconds in the presession-shuttling mode. Spectrum analyses of the recorded vibration data showed that the gearbox operated normally through the 15 second start-up and for 45 seconds at full speed before precession-shuttling started. After precession-shuttling started, the vibration of the unloaded end of the pinion (opposite coupling) became very severe. The spectrum analyses showed many strong harmonics of running speed vibration indicating impacts or orbiting of the

TABLE 1

Gear Data	
Overall Dimensions	L (77 Inches), W (38 Inches), H (62 Inches)
Transmitted Horsepower	6900 (Includes Motor Service Factor)
Type	Double Helical
AGMA Service Factor	1.57
"K" Factor	116
High-Speed Shaft rpm	9302
Low-Speed Shaft rpm	1780
Gear Ratio	5.225/1
Rotation	Down Toward Pinion (Down Mesh)
Shaft Centerline Distance	27.250 Inches
Radial Bearings	Plain Sleeve
Thrust Bearing	Double Tapered Land, On Gear Shaft
Pinion Journal Diameter	4 Inches
Gear Journal Diameter	7 Inches
Lubrication	Spray Feed into Mesh
Bearing Lube Pressure	10 Pounds Per Square Inch Gauge
Oil	Shell LO Hydrax 32, 150 SSU at 100°F, 32 Centistokes at 40°C
Exclusion Pan	Lower Half of Gear, No Oil Level in Gear-Box
Overall Face Width	15.25 Inches
Active Face Width	12.50 Inches
Helix Angle	34.02 Degrees
Pressure Angle	19.66 Degrees
Tooth Loading	854 Pounds Per Inch of Tooth Face
Pitch-Line Velocity	21,318 Feet Per Minute
Pitch Diameter, Gear	45.74 Inches
Pitch Diameter, Pinion	8.75 Inches
Teeth on Pinion	31
Teeth on Gear	162
Finish	Hob and Shave
Backlash	0.010 Inch
Diametral Pitch	3.54
Material, Gear and Pinion	AISI 4340
Pinion Hardness	270 Brinell
Gear Hardness	215 Brinell

pinion in loose fitting bearings. The vibration diagnosis was precession-shuttling caused by the overhanging weight of the coupling or a large unbalance couple.

A third trial run was made after reinspecting the gears and bearings for damage and removing the coupling half. The pinion had two 180-degree opposed keyways; thus it was necessary to install a half-key. The gear-box ran perfectly with insignificant vibration, a low noise level, and no signs of precession-shuttling of the pinion.

DETAILED INSPECTION

The next operation was to inspect various parts of the gear-box in detail to try to confirm the cause of the noise and vibration and to determine a course of action for repair. Table 2 is a list of items inspected to determine the condition of the gear-box.

REPAIR CONSIDERATIONS

Previous experience with high-speed couplings indicated that the 3.5-mil hub-to-shroud clearance would close up to about 2.5 mils when the coupling hub was shrunk on the shaft. Since no coupling unbalance had been experienced with similar clearances in high-speed couplings before significant tooth wear had occurred and since the balance had been found excellent, thoughts of coupling unbalance due to eccentricity were abandoned. Coupling weight reduction was abandoned because precession-shuttling occurred with only one hub mounted. Lighter couplings, that were within our selection capability at that time, would not have reduced the hub weight very much.

Addition of a weight on the opposite end of the pinion shaft to balance the weight of the coupling was considered, but the idea was abandoned for personnel safety reasons. There seemed to be no really safe way to connect a weight to the free end of the pinion shaft. Abandoning this idea was fortunate because some recent experience with high-speed pinions having large bearing clearances and an overhanging on each end has been rather unsatisfactory.

Relocation of the spray lubrication from upstream of the gear mesh to downstream of the mesh was considered but dropped because the gears ran very well without a coupling in place.

The pinion had not yet experienced misalignment forces; however, good alignment became a top priority.

The excessive pinion bearing clearances of the following two locations, (1) journal to bearing inside diameter and (2) bearing outside diameter to bearing housing inside diameter, were the most significant findings; therefore, controlling bearing clearances became the target for corrective action. A set of bearings was selected with 8 mils clearance and a 1/32-inch deep pressure dam 3 inches wide and 135 degrees long was machined from 3:00 o'clock to 10:30 o'clock in the top bearing half (see Figure 3). When the bearings were installed, the horizontal joint of the bearing insert was shimmed and doweled to make the bore concentric and parallel to the outside diameter, and shims were added between the top of the bearing insert and the inside diameter of the bearing housing to

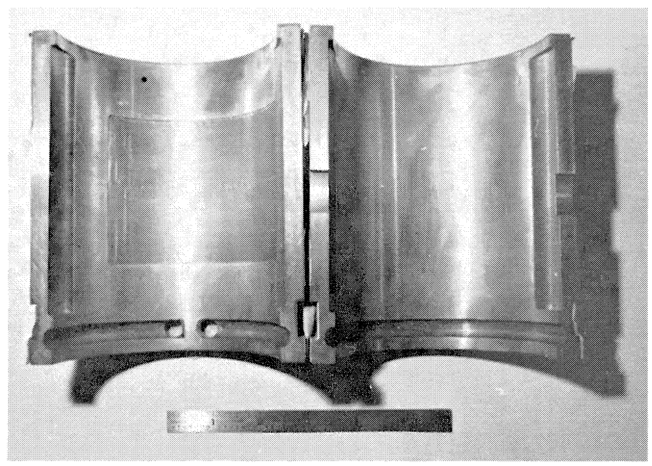

Figure 3. Modified Pinion Bearing Pressure Dam was Cut and Shims were Used at Horizontal Joint.

TABLE 2

Inspection of Gear-Box

The following items were checked:

Item	Condition
1. Tooth Contact	Very good.
2. Pinion Balance	Excellent.
3. Bearing Clearance	The bearing clearance was 10 mils on a 4-inch shaft.
4. Apex Runout	Measurement capability here was poor. Nothing significant was found. Sides of gear and pinion were true. Pinion did not have axial runout relative to gear. (Apex runout is the only cause for pinion shuttling that was cited in the literature surveyed in an effort to diagnose the problem.)
5. Teeth	Pinion and gear teeth were found very lightly scuffed on inactive side indicating large axial motion and impact loading.
6. Coupling Balance	Each piece of the coupling and the complete coupling assembly were check balanced. The balance was excellent.
7. Keys	Keys were found to be weight matched.
8. Coupling Hub-to-Shroud Clearance	Radial pilot clearance between outside diameter of hub teeth and inside diameter of shroud teeth roots was 3-½ mils which is professed by coupling manufacturers to be too large.
9. Bearings	The bearings were not bored parallel to the centerline. During manufacture, the babbitted bearing insert was machined round and and sawed in half leaving a clearance approaching 10 mils between the top half of the bearing and the bearing housing.

assure a good fit. The spots were scraped from the inside diameter of the bearings. Some corrective scraping was done to improve tooth contact, and a final tooth contact correction was made by shimming one corner of the gear-box.

When the gear-box was test-operated after modification, it showed only modest signs of precession-shuttling which we thought would disappear when load was added. The compressor train was successfully test operated with helium in the antisurge loop, and it was placed in process service when the plant was started. Operation of the gear in the longer term appeared normal except for bearing effluent oil temperatures which reached 180°F, about a 50°F to 60°F temperature rise.

EVALUATION OF VIBRATION

The gear-box was vibration monitored routinely using accelerometers mounted on the bearing housings. The vibration spectrum analyses appeared normal except for the mesh frequency, 4806 Hz, which was about 20g.

$$\text{Mesh Frequency} = \frac{\text{Teeth} \times \text{RPM}}{60} = \frac{31 \times 9302}{60} = 4806 \text{ Hz} \tag{1}$$

This vibration was high enough to warrant a statistical look at gear vibration (see Figure 4). The statistical spectrum presented was assembled from a five-year accumulation of vibration data in our files for 22 comparable high-speed gears. When the 20g mesh-frequency vibration of this gear was removed from the sample, the next highest vibration was 10g. Ninety percent of the gears had mesh-frequency vibration of 3g or less. The median mesh-frequency vibration was 1.3g.

Examination of the data from several gear failures led to the conclusion that a 10g mesh-frequency vibration was operable; 20g was unpredictable; and 30g to 40g was very close to inoperable. Gears with visible damage to the tooth finish vibrating 30g to 40g would not last more than a few days.

As a result of this vibration experience, we became overly enthusiastic during one maintenance shutdown and reduced the pinion bearing clearance from 8 mils to 6 mils on the 4-inch journal. The mesh-frequency vibration was reduced to 10g but the bearing ran too hot (200°F). After several weeks, we were saved by failure of another piece of equipment which caused a plant shutdown, allowing us to restore the 8-mil pinion bearing clearance. Careful observation of mesh-frequency vibrations and bearing clearances from 1970 to 1975 led to a firm conclusion that pinion precession-shuttling could be controlled by reduced bearing clearances if we could solve the bearing overheating problem. Over this time span, the mesh-frequency vibration varied from 10g to 30g while the bearing clearance varied from 6 mils to 9 mils. Operating time at 30g mesh-frequency vibration was very limited.

DECISION TO MODIFY BEARINGS

Several unfavorable observations and incidents led to a decision to modify the pinion bearings:

1. Mesh-frequency vibrations were statistically high and very sensitive to bearing clearance (workmanship) in spite of very close engineering supervision.
2. Inspection of the gear at various maintenance shutdowns revealed that pinion performance had not consistently been as good as originally anticipated. Axial pinion motion of at least 3/32 inch at some time during operation had made a 1/32-inch deep imprint of the pinion shaft end in the face of the coupling spacer. The imprint was made by repeated blows, but the surface was not fatigue failed by persistent vibratory motion.
3. During 1975, the plant was shut down twice by overheated pinion bearings caused by contaminated oil.

Sketches had been made of the pinion bearing housing; bearing housing thickness had been measured; and consultation with a bearing design shop had established that the pinion bearing housings could be rebored to a much larger diameter and fitted with tilting-shoe bearings in a reasonable time span and for a reasonable price. Our experience and the bearing designer's experience indicated that we could expect these bearings to run cool with a 6-mil clearance.

During the November 1975 maintenance shutdown, the gear-box was sent to the bearing shop where the pinion bearing housings were rebored and fitted with five-pad, tilting-shoe bearings. The pads of thin-babbitted bronze were mounted on hemispherical buttons giving them freedom in two planes (see Figure 5). When the gear-box was reinstalled at the job site, it was found that the 6-mil radial clearances were correct, the

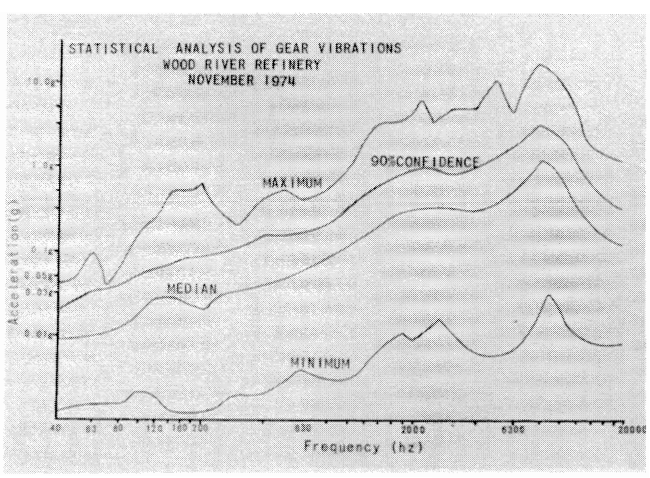

Figure 4. Statistical Vibration of High Speed Gear Boxes. Accelerometer Data Taken from Bearing Housings of 22 Gear Boxes with Mesh Frequencies from 3000 Hz to 10,000 Hz.

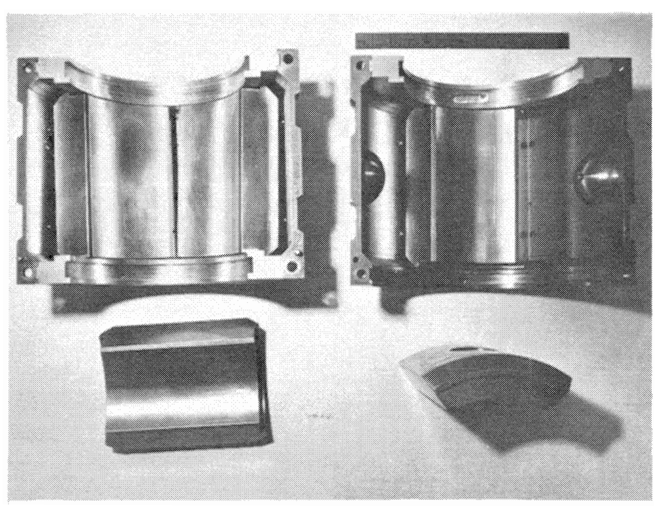

Figure 5. Tilting Shoe Radial Bearings with Six Mil Clearance before Installation on Pinion Shaft.

bearing shells were an excellent fit in the bearing housing, and only very minor adjustments to the gear-box shims were required to obtain a very good tooth contact.

OPERATION AFTER MODIFICATION

The gear-box, when placed in service, ran beautifully with only a 15°F rise in bearing oil temperature. The mesh-frequency vibrations at 4806 Hz dropped from 26g to 3.1g (see Figure 6). The mesh-frequency noise dropped 12dB from 94dB to 82dB when measured in the near field (see Figure 7). The noise measurement 3 feet to 4 feet from the machine dropped 1dB to 2dB. This small general reduction was caused by the sound power of other contributing noise sources. Near-field measurements were made 2 inches to 3 inches from the gear-box cover just above the horizontal joint. In this noisy environment, near-field measurements were the only way to be sure gear noises were received exclusively.

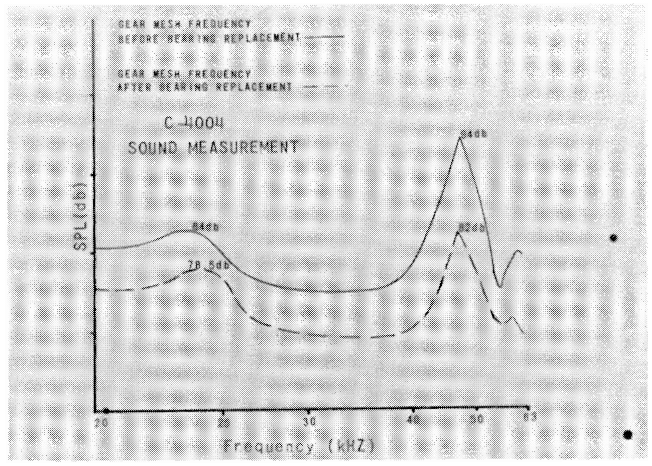

Figure 7. Noise Spectrum Analyses Showing Reduction of Mesh Frequency Noise in the Near Field from 94 dB to 82 dB.

FURTHER NOISE INVESTIGATION

In the course of making sound measurements, a strong noise peak of 105dB was noticed at 1200 Hz (see Figure 8). Very rough wave length calculations indicate that the width of the gear-box in the large space above the pinion is two acoustic wave lengths at 1200 Hz, in air.

$$\text{Wave Length} = \frac{\text{Velocity of Sound}}{\text{Frequency}} = \frac{1100\ \text{Ft/Sec}}{1200\ \text{Hz}} \times 12 = 11\ \text{Inches} \tag{2}$$

We presume the source of excitation was windage from the top half of the gear. The oil suspended in the air makes it impossible to know the true speed of sound in the gear-box; therefore, the plan is to drill and tap a hole in the side of the gear-box cover above the pinion and insert a microphone to get a sound pressure profile across the width of the box. If this experiment proves the existence of a standing wave, an installation of baffles at pressure nodes (velocity antinodes) will be made in the top half of the box over the pinion to break up the standing wave pattern. An alternative would be to install Helmholtz chambers at pressure antinodes to interfere destructively with the standing wave. A howling pump and a howling boiler superheat header have been successfully silenced with these resonant cavities. Helmoholtz chambers must be pretuned experimentally, and they must be kept the same temperature as the flowing fluid to avoid detuning.

To round out the hunt for noise problems, this particular gear-box has been examined for "sound-windows." This is a condition where a gear-box's mechanical natural frequency matches a noise frequency. The sound pressure in the gear-box causes the resonant panel of the box to vibrate freely transducing the noise to the atmosphere with very little attenuation. This is a common defect in gear-boxes as well as other machines. Sound-windows can be corrected by stiffening the panel or, where practical, by tying together, internally, the two sides of the box which vibrate 180 degrees out of phase.

The technique for finding sound-windows is to plot vibration response of the gear-box versus frequency. A plastic hammer can be used to shock or excite one box panel at a time while response is measured. A real time analyzer or an automated vibration versus frequency plotter receiving the output of a vibration velocity transducer will give a good response curve. The response curve and a narrow-band frequency analysis of the near-field noise are then compared to find high response frequencies matching noise frequencies. Box rein-

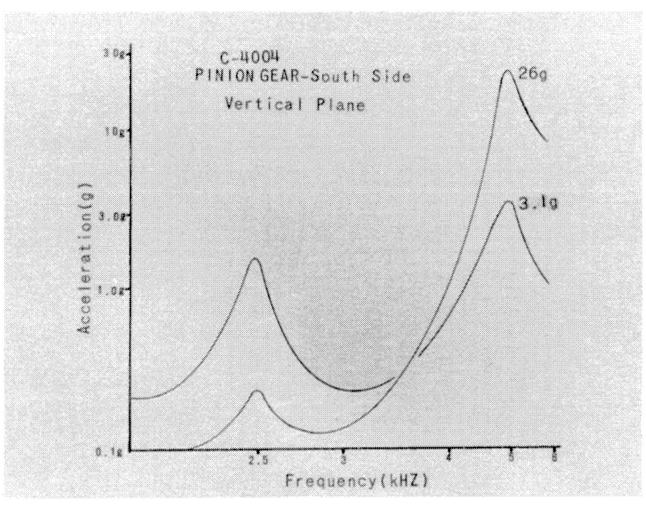

Figure 6. Vibration Spectrum Analyses Showing Reduction of Mesh Frequency Vibration from 26g to 3.1g at 4806 Hz.

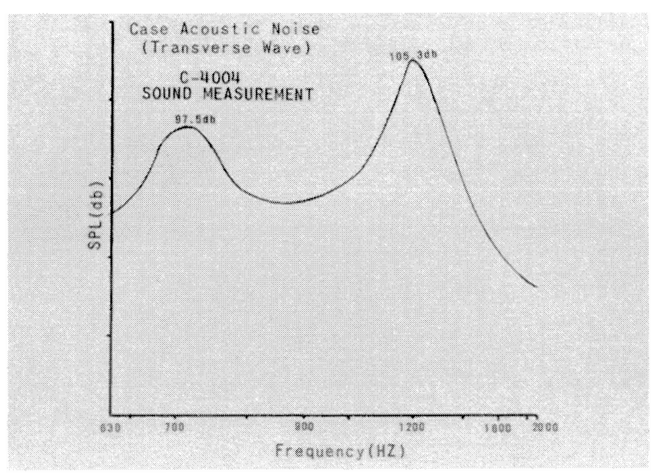

Figure 8. Noise Spectrum at 1200 Hz. Inside Width of Rectangular Box Cover above Pinion is Exactly Two Wave Lengths, 22 Inches.

forcement can then be applied carefully to these suspect areas. The box should be retested after application of the reinforcement to be sure the natural frequency has been changed as anticipated. This particular gear-box has a number of natural frequencies in the 200 Hz to 400 Hz range; fortunately, there was no exciting noise in this frequency range. Vibration excitation has also been very small in this frequency band.

CONCLUSION

Pinion precession-shuttling problems do not occur very often; but when they do, the gear purchaser is highly inconvenienced. The gear is only a small part of a plant that generates many thousands of dollars per day in revenue. The user could easily afford to buy another gear if the one on hand could be made to work temporarily. However, there is no real assurance that the second gear would not have the same precession-shuttling problem unless cool-operating, tilting-shoe bearings with small clearances (1-½ mils per inch of shaft diameter) are installed. Thin babbitted bronze or copper-backed tilting-shoe bearings could be installed in the malfunctioning gear if the bearing housing is thick enough to be rebored. Gears have high bearing loads, enough so that the bearings installed on our subject gear were marginal enough to require redesign. The original tilting-shoe bearings are operating and have caused no outage. Needless to say, the responsibility for this kind of equipment alteration is high, and it belongs solely with the equipment user. Anyone attempting to make such a change should obviously have contingency plans in mind or partially implemented.

Mesh-frequency vibrations are an excellent way to forecast gear problems by deviation from statistical criteria and by deviation from original "as-installed" data. A gear not previously mentioned suddenly acquired 40g mesh-frequency vibration at 7623 Hz. A disturbance of this kind is audible but seldom recognized. Vibrations at these high frequencies are not sensible to touch. The plant was shut down, and the gear and couplings were examined. The high-speed, continuously-lubricated coupling was fouled with deposits increasing misalignment forces which caused the pinion to orbit, producing excessive mesh-frequency vibration.

Noise levels can be improved in the design stage by specifying and getting close-clearance tilting-shoe pinion bearings, by baffling the box to prevent acoustic disturbances, and by buying a heavy box not subject to mechanical resonances in the audible frequency range. The heavy box is a good attenuator, and the lack of mechanical resonances at audible frequencies prevents the sound-window problem.

The conventional parameters should not be forgotten. Good tooth finish, small teeth, light tooth loading, proper tooth hardness, etc., all contribute to a quiet, long-life gear.